10/501,148
07/13/04

CHARTER MARK

PATENTS • DESIGNS • The Patent Office • COPYRIGHT • TRADE MARKS

PCT/GB 2003 / 000194

INVESTOR IN PEOPLE

The Patent Office
Concept House
Cardiff Road
Newport
South Wales
NP10 8QQ

PRIORITY DOCUMENT
SUBMITTED OR TRANSMITTED IN COMPLIANCE WITH RULE 17.1(a) OR (b)

REC'D 28 MAR 2003
WIPO PCT

I, the undersigned, being an officer duly authorised in accordance with Section 74(1) and (4) of the Deregulation & Contracting Out Act 1994, to sign and issue certificates on behalf of the Comptroller-General, hereby certify that annexed hereto is a true copy of the documents as originally filed in connection with the patent application identified therein.

In accordance with the Patents (Companies Re-registration) Rules 1982, if a company named in this certificate and any accompanying documents has re-registered under the Companies Act 1980 with the same name as that with which it was registered immediately before re-registration save for the substitution as, or inclusion as, the last part of the name of the words "public limited company" or their equivalents in Welsh, references to the name of the company in this certificate and any accompanying documents shall be treated as references to the name with which it is so re-registered.

In accordance with the rules, the words "public limited company" may be replaced by p.l.c., plc, P.L.C. or PLC.

Re-registration under the Companies Act does not constitute a new legal entity but merely subjects the company to certain additional company law rules.

Signed A. Evans.

Dated 13 March 2003

tents Form 1/77

Patents Act 1977
(Rule 16)

K
17 JAN 2002
NEWPORT

1/77

17JAN02 E688560-1 C81053
P01/7700 0.00-0200991.8

Request for grant of a patent

(See the notes on the back of this form. You can also get an explanatory leaflet from the Patent Office to help you fill in this form)

17 JAN 2002

The Patent Office

Cardiff Road
Newport
South Wales
NP10 8QQ

1. Your reference

   EATPAT3

2. Patent application number
   (The Patent Office will fill in this part)

   0200991.8

3. Full name, address and postcode of the or of each applicant *(underline all surnames)*

   E. A. TECHNICAL SERVICES Ltd
   9 RYDAL PLACE
   CLITHEROE ROAD
   LANCASHIRE BB74JY
   ENGLAND

   Patents ADP number *(if you know it)*

   83064774001

   If the applicant is a corporate body, give the country/state of its incorporation

4. Title of the invention

   COMPRESSOR WITH VARIABLE PRESSURE AND FLOW CONTROL

5. Name of your agent *(if you have one)*

   "Address for service" in the United Kingdom to which all correspondence should be sent *(including the postcode)*

   E. A. TECHNICAL SERVICES Ltd
   9 RYDAL PLACE
   CLITHEROE ROAD
   CHATBURN
   LANCASHIRE BB74JY

   Patents ADP number *(if you know it)*

6. If you are declaring priority from one or more earlier patent applications, give the country and the date of filing of the or of each of these earlier applications and *(if you know it)* the or each application number

Country	Priority application number *(if you know it)*	Date of filing *(day / month / year)*

7. If this application is divided or otherwise derived from an earlier UK application, give the number and the filing date of the earlier application

Number of earlier application	Date of filing *(day / month / year)*

8. Is a statement of inventorship and of right to grant of a patent required in support of this request? *(Answer 'Yes' if:*
   a) any applicant named in part 3 is not an inventor, or
   b) there is an inventor who is not named as an applicant, or
   c) any named applicant is a corporate body.
   See note (d))

   YES

9. Enter the number of sheets for any of the following items you are filing with this form. Do not count copies of the same document

Continuation sheets of this form	14
Description	7
Claim(s)	0
Abstract	0
Drawing(s)	7 ONLY

10. If you are also filing any of the following, state how many against each item.

Priority documents	
Translations of priority documents	
Statement of inventorship and right to grant of a patent *(Patents Form 7/77)*	
Request for preliminary examination and search *(Patents Form 9/77)*	
Request for substantive examination *(Patents Form 10/77)*	
Any other documents *(please specify)*	

11. I/We request the grant of a patent on the basis of this application.

Signature E A Driver Date 16 JAN 02

12. Name and daytime telephone number of person to contact in the United Kingdom RON DRIVER 01200 441492

Warning

After an application for a patent has been filed, the Comptroller of the Patent Office will consider whether publication or communication of the invention should be prohibited or restricted under Section 22 of the Patents Act 1977. You will be informed if it is necessary to prohibit or restrict your invention in this way. Furthermore, if you live in the United Kingdom, Section 23 of the Patents Act 1977 stops you from applying for a patent abroad without first getting written permission from the Patent Office unless an application has been filed at least 6 weeks beforehand in the United Kingdom for a patent for the same invention and either no direction prohibiting publication or communication has been given, or any such direction has been revoked.

Notes

a) *If you need help to fill in this form or you have any questions, please contact the Patent Office on 08459 500505.*

b) *Write your answers in capital letters using black ink or you may type them.*

c) *If there is not enough space for all the relevant details on any part of this form, please continue on a separate sheet of paper and write "see continuation sheet" in the relevant part(s). Any continuation sheet should be attached to this form.*

d) *If you have answered 'Yes' Patents Form 7/77 will need to be filed.*

e) *Once you have filled in the form you must remember to sign and date it.*

f) *For details of the fee and ways to pay please contact the Patent Office.*

Compressor with Variable Pressure and Flow Control

Spark ignition engines conventionally control their power output by controlling the amount of air that passes through the intake system. A throttle regulates the airflow and at maximum power the throttle is fully open and at idle the throttle is substantially closed. When the throttle is partly closed the engine's intake manifold is below ambient air pressure and the engine has to do work to draw in the air.
The maximum temperature at the end of the compression stroke in a spark ignition engine is limited by the need for satisfactory combustion and combustion timing, the maximum compression temperature can easily be reached with conventional compression ratios. When an engine is supercharged the efficiency of compression by the supercharger is generally less than the engine's compression efficiency and this results in a higher temperature for a given pressure than a naturally aspirated engine on its own. A supercharged engine normally has the supercharged air cooled in a heat exchanger and the compression temperature limit generally necessitates a lowering of the engine's compression ratio. With a supercharged and reduced compression ratio engine the pressure at the end of the power stroke is higher than in a naturally aspirated engine and to reduce the waste of this energy the exhausted gasses are generally passed through a turbine.
Private automotive vehicles spend most of their time at part power and in the case of spark ignition engines this means at part throttle with its attendant throttling losses. It is known in the art that improvements in engine efficiency could be made if the throttling losses could be eliminated.
There are two ways of eliminating throttling losses; one is to recover the losses by putting a turbine in the intake and the other is to eliminate the throttling process by not having any part of the cycle at pressures below ambient. To achieve the latter and have an acceptable range of power means an engine at idle must have:

- A cylinder full of air at ambient pressure
- Have a reduced compression ratio
- Gradually increase the amount of pressure ratio until maximum power is reached.

It will be appreciated that since conventional engines have their cylinders full of ambient pressure air at full power, to have an engine with a cylinder full of ambient air at low or idle speed, and yet only give idle power, means the engine must be appreciably smaller for the same low power requirement. Or alternatively, the air flow must be regulated.

Regulating the flow and degree of supercharged air entering an engine has been difficult and inefficient. It has been difficult because superchargers could not control the degree of supercharge accurately enough over the required range and inefficient because compression efficiency and airflow control was poor and wasteful.
In spark ignition engines combustion will only take place within very narrow limits of fuel to air ratio. Gasoline Direct Injection (GDI) is used to provide specific regions of the engine's cylinder with a combustible mixture, whilst allowing other regions to have an increased proportion of air, thus reducing the amount of throttling required. Another method of eliminating throttling losses is to vary the valve timing and valve lift (VVT), this allows some of the air that entered the cylinder to be pushed out by the piston before the valves close. Both GDI and particularly VVT increase the cost and complexity of engines.

Over the past few years hybrid engines have been proposed that were a combination of electric motor and a relatively small engine running at near maximum power whenever it was used. More recently there has been a move to a higher voltage electrical system, this permits engines to stop when the vehicle stops and then for the vehicle to initially move off using the electric motor.

In the present invention it is proposed to use a combination of supercharger, internal combustion engine and exhaust turbine. The exhaust turbine may drive a compressor or electrical generator or both. The enabling technology to permit efficient use of this combination of components is the use of a supercharger of the type and incorporating the features described in application PCT/GB01/03089 and the features described in the present invention. This type of supercharger allows the internal combustion engine's airflow to be controlled. It takes a full charge of air each revolution and evacuates air not required by pushing it out through the side disc metering orifice or orifices and allows the remainder to be discharged to the engine. If sufficient air is evacuated and the volume remaining is less than is required to fill the engine's cylinder with ambient pressure air, the cylinder pressure will fall to below ambient and with it the pressure on the outlet side of the supercharger. The difference between ambient air pressure and the supercharger outlet will drive the supercharger, thus recovering the energy used by the engine to produce the partial vacuum in the cylinder (the throttling losses). In this manner the supercharger can supply air from below ambient pressure to maximum supercharge pressure. This type of supercharger has compression efficiency comparable with the efficiency of the compression within an engine and an ability to accurately control airflow. This combination of components eliminates the need for expensive GDI and VVT systems and with the exception of the supercharger needs only conventional components and fuel systems although using GDI may increase the range of power. Adding a heat exchanger to the combination enables an engine of about 1-litre to have the same power output as a 2-litre engine but with a considerably reduced weight and fuel consumption.
With the swept volume of the internal combustion engine known, a supercharger of this type can be designed for a particular supercharger maximum pressure and with the inlet control the supercharger output pressure can be varied from below ambient to maximum pressure. Under these conditions the supercharger's outlet orifice or orifices position and size are constant and no variation is necessary.
Control of either inlet or outlet is simply achieved by exposing more or less orifice area. Having apertures in the rotor disc, casing and an outer ring most easily does this. By sliding the outer ring over the interposed casing, more or less casing apertures are exposed, when the rotor disc apertures are adjacent the exposed casing apertures air can pass through if the position of the slide allows it. By this method pressure and mass flow can be controlled.
With the impending widespread introduction of higher voltage electrical systems in vehicles, auxiliary equipment will increasingly be driven by electric motors rather than directly by the internal combustion engine. Using an electric motor and varying the machine speed relative to the engine speed could additionally control the airflow in the present invention.

Similar machines can be used for compressing other fluids, for instance refrigerants. Machines that compress refrigerants are normally referred to as heat pumps. Heat pumps generally run at constant speed and stopping and starting the machine several times over a period of time normally controls the average heat output. By using a

sliding ring to vary the exposed orifice size and position in heat pumps it is possible to vary the pressure and heat output of them. By adding a variable speed motor a full range of heating and cooling outputs can be achieved without "stopping and starting" the machine. Heat pumps below a certain heat output don't or aren't able to recover the energy available in the fluid returned from the condenser. In PCT/GB01/03089 referred to above, it described how energy can be recovered in a machine of this design. A further consequence of the control system using the sliding ring described here is the ability to control the inlet conditions to the expander by controlling the compressor outlet and/or inlet conditions. Thus not only can the parameters of speed, pressure and heat output from the compressor be controlled; the expansion inlet conditions for the turbine expander can be controlled indirectly by controlling one or a number of compressor outlet parameters. It will be obvious that a sliding ring at the expander inlet could also be used for expander inlet control. The benefits though, of controlling the expander inlet may be outweighed by the loss in fluid properties caused by the sudden expansion or "flashing" of the refrigerant. There may therefore be substantial benefits of indirectly controlling the flow by controlling the compressor flow, and using a constant expansion orifice and a wide piston rotor side disc to create a gradually increasing orifice area for a smooth expansion.
In the present rotary or rolling piston compressor the volume of fluid trapped between the piston and the vane can be varied from a full to minimum charge by allowing the charge to pass out again through the orifices before compression begins. In the case of a supercharger the full charge would be the amount required to fill the internal combustion engine's cylinder with air at the design pressure and the minimum would be the volume required to fill the cylinder at either ambient pressure or, if a partial vacuum was required, to the volume required for that pressure.
By having the casing interposed between the slide and the rotor, fluid can pass into or out of the machine when orifices in all three components are aligned.
To maintain high thermodynamic efficiency, manufacturing clearances need to be of the order of 0.02mm, which leaves very little allowable distortion due to centrifugal, inertial or fluid pressure forces. The pressure forces are approximately 2 Bar for superchargers, 3 Bar for fuel cell compressors and ranging from 15 to 90 Bar for heat pumps.

Rolling piston compressors with either sliding, hinging or swinging vanes have been known in the art for over 100 years. . All machines of this type to date have not been able to efficiently vary the outlet pressure nor have they been able to work as a turbine without valves and complex valve timing. Those with sliding vanes were limited by cantilever forces, speed and friction, those with hinging vanes were limited by deflection of the material and fatigue life, swinging vanes are limited by friction. Superchargers for internal combustion engines cannot have oil lubrication for environmental reasons, heat pumps can have oil mixed with the refrigerant but this degrades and reduces the heat exchanger efficiency. This leaves hinging vanes as the most likely candidate for automotive use, and the one worthy of further consideration for heat pumps.

In automotive applications the minimum maximum speed is likely to be 6000 revolutions per minute and for heat pumps 3600 revolutions per minute. The speed and size of automotive applications gives rise to inertia loads and heat pumps have high pressure loads. Both these loads cause deflection of conventional vane shapes

and the increased clearances to allow for the deflection causes a loss of efficiency by allowing fluid to leak away.
In the present case the vane is shaped to give maximum resistance to deflections due to inertia and pressure. The shape and positioning of the vane relative to the casing and actuating mechanism to minimise the loads means there would be a restriction on the area for fluid flow under the vane and into the machine. If the circumferential length of the inlet orifice was increased to overcome the inlet flow restriction the machine's capacity would be reduced. Allowing the fluid to enter through the vane has eliminated the inlet flow restriction. The vane shape therefore provides maximum resistance to inertia and pressure loads and minimum resistance to inlet fluid flow.

When the machine is used as a supercharger the volume flow is fixed by the physical size of the internal combustion engine's cylinders but the mass flow is determined by the engine's power requirement. Varying the supercharger outlet pressure can vary the mass flow. In the present case the supercharger is initially filled with ambient air and as the piston rotates towards the vane the air is compressed. The mass of air to be compressed can be varied and therefore its pressure, by allowing some of the air to be evacuated before compression begins. This is achieved by providing orifices in the sides of the rotating piston and in the casing. When a slide exposes holes in the casing the air can flow out of the machine through the piston side and casing holes. The slight air pressure rise as the piston advances towards the vane provides the pressure drop needed to evacuate the air.
It is unlikely that the outlet volume of air will need to be varied, therefore there will be no need for a slide on the outlet and the outlet holes can be fixed and the compressed air discharged as described in PCT/GB01/03089. However should there be a need to vary the outlet volume orifice area a slide can be provided in a similar manner to that provided for evacuation as described earlier.

Control means described herein have inherent inefficiencies caused by the volume of fluid retained in the orifices of the piston side discs and casing apertures and then transferred from high to low pressure regions as the rotor turns.

By making the side discs thin in an axial direction and transferring the fluid medium axially as shown in US 3895609, the transferred fluid is small but leakage around the orifice and over the periphery of the side disc will increase as the axial length of the disc decreases. Fitting a contacting seal to reduce this leakage would cause friction and therefore loss of machine efficiency and eventual wear would provide losses from this leakage. This axial exit design produces a high rotor end load when the plain side disc face covers the exposed casing holes. Equalising the end load by providing transfer holes from one side disc to the other allows leakage over both discs and back into the machine under supercharge conditions, and out of the machine under vacuum conditions. If the side disc width is increased the fluid transferred from high pressure to low pressure is increased as the rotor turns.
In the case of PCT/GB01/03089 the side discs are relatively long in the axial direction and leakage is governed by pressure drop, radial clearance, surface roughness, axial length and fluid properties. Leakage in this case is relatively insensitive to changes in axial length and to radial clearance changes caused by changes in machine temperature and there is no end load caused by pressure. With this radial exit design

the rotor transfer orifice volume increases as the disc width increases and with it the transfer losses.
By making the side discs in US 3895609 "L" shaped and bringing the casing round the contour of the "L", the leakage over the discs can be reduced. The side disc running clearance and the working volume are in direct communication with each other via the disc holes, thus two leakage paths are available for the fluid. By interposing the casing between a slide and the rotor and fitting a face slide control, the flow and pressure can be controlled in a similar manner to that described above.

As with all machines of this type their efficiency is crucially dependant on the amount of leakage, contact seals will produce unacceptably high friction losses apart from those on very large machines where the percentage of friction can be brought within acceptable bounds.
With axial entry or exit the change in axial clearance due to thermal effects will affect machine efficiency and this will impose a limit on the machine's length. This together with the difficulty of accurately machining several parts and assembling them and a bearing together, and the required total clearance gap of less than 0.02mm, makes a satisfactory design difficult.
With radial exit and entry the manufacturing of two round components is easily controlled. A manufacturing easement for concentrically fitted parts could be to put a lining or abradable coating on one component, for instance a polymer material, which would allow wear on the first rotation, then the maximum clearance obtained is that caused by thermal expansion.

With an axial entry and exit and face slide control there is a need to hold the slide in position, rotate it, and to fully react any tendency for pressure to lift it off the face. In the case of a radial slide, the diameter of a slide closely fitted to the diameter of a casing automatically reacts any pressure loads and therefore only requires to be rotated.

In the present invention air or fluid exits the machine when orifices are in alignment as the machine rotates. There are no conventional valves and manufacturing costs are reduced and reliability increased. There are penalties for this simplistic design though. Pushing the fluid out of the machine produces a pressure drop across the orifice, and the extra pressure is a parasitic loss. By increasing the orifice area this loss is reduced but the loss from transferring fluid from high to low pressure increases. In the case of radial exit with little pressure loss the loss from transferred fluid is 10%. A compromise between pressure and transfer losses results in an overall loss of 5%. With axial exit the transfer loss is small but the leakage is high, adding an axial extension to the rim of the disc reduces the leakage but accurate axial clearance control is required to produce an overall loss of 5%. The added disadvantage of the increased cost of a slide control makes this design unattractive. However in the case of outlet to the internal combustion engine when there is no requirement for slide control there may be some vehicle installation requirements that make axial exit desirable. Therefore a combination of axial and radial exit and evacuation may be desirable under some circumstances.

The invention may be performed in various ways and some specific embodiments will now be described by way of example with reference to the accompanying diagrammatic drawings, in which:

Fig 1 shows a vane and its lever arm;
Fig 2 shows a vane in its position in the casing;
Fig 3 shows the piston rotor;
Fig 4 shows the whole machine with a cover and end casing removed;
Fig 5 shows a different view of Fig 4;
Fig 6 shows an end view of Fig 4;
Fig 7 shows a piston rotor with radial evacuation to one side and axial exit to the other;
Fig 8 shows a different view of Fig 7.
Figures 9 to 14 shows the fluid flow path for the different configurations, in which:
Fig 9 and Fig 10 shows a piston rotor with axial exit and radial evacuation;
Fig 11 and Fig 12 shows a piston rotor with radial exit and evacuation;
Fig 13 and Fig 14 shows a piston rotor with axial exit and evacuation.

The invention is based on a rotary positive displacement machine comprising:

- A stator having a circular cylindrical internal surface delimiting an operating chamber, the stator having fluid inlet and outlet ports;
- A rotor in the operation chamber, the rotor being mounted so as to be rotatable relative to the stator about the axis of the said internal surface, the rotor having a cylindrical external surface, the said axis passing through the rotor, a generatrix of the external surface being adjacent to the said internal surface, and a diametrically opposite generatrix being spaced from the said internal surface;
- A sealing member projecting substantially radially through a slot in the stator into the operating chamber and being movable substantially radially, the sealing member extending parallel to the said axis and having a length substantially equal to that of the rotor;
- Means for keeping the radially inner end of the sealing member adjacent the internal surface of the rotor;
- Means for varying the number of fluid outlet ports;
- Means for varying the fluid mass flow.

In Fig 1 one of the vane's profiles is concentric with the vane's pivot axis and is a sealing face with a static conforming face of the casing. The flat sides of the vane are also sealing faces with the casing. Fig 1 also shows a lever arm which may be integral or attached to the vane. Optional weight reduction holes are also shown.
Fig 2 shows the relationship of exit and evacuation holes in the casing to the vane and to the rotor direction of rotation. The rotor is not shown. The fluid inlet slot typically extends 40 degrees from the point at which the vane profile intersects the casing inner surface (0 degrees) and the region for evacuation a further 140 degrees. The exit region is typically from 240 to 360 degrees. The number of degrees in each region will vary as the machine size varies and on the optimisation of fluid pressure loss and fluid transfer loss.
Fig 3 shows pictorially a rotor with one type of exit and evacuation slot in the piston rotor side discs. To equalize pressure acting on both sides of the rotor and prevent axial end loads a small fluid transfer hole is provided. To minimise the volume of fluid that could be held inside the rotor the large weight reduction hole and any other weight or material reduction features will be filled with a cheap lightweight material or made hollow.

Fig 4 shows the fluid path into the machine through the vane. The slide ring for controlling evacuation is shown and several of the fixed exit holes.
Fig 5 shows the slide ring and some of the evacuation holes.
Fig 6 shows a typical attachment of lever arm and connecting rod.
Fig 7 and Fig 8 shows two view of the rotor modified to accommodate an axial exit to one side of the rotor through an "L" shaped side disc.

Machines similar to the present invention have been used for many years but have been limited by high friction or poor efficiency or both. The machine in the present invention is required to have an efficiency of compression and expansion of better the 80% and virtually no friction. To achieve this level of performance running clearances need to be of the order of 0.02mm and to be practical it need to be capable of mass production. It will be obvious that increasing the running clearance to 0.03mm will increase the flow area by 50% but because of viscous effects at these small clearances the leakage flow rate will increase by more than 50% to perhaps 100%. By the same reasoning doubling the length of the flow path can reduce the flow by more than half. To be able to reliably manufacture on a mass production basic machines requiring this degree of accuracy, means the method of manufacture and assembly needs to be considered.
It is relatively easy to make two parts with conforming diameters and thermal affects on running clearance are proportional to the radius. On the other hand axial clearance –particularly when bearing axial location is taken into account- is much more difficult, and thermal expansion is proportional to length. Provided the basic machine efficiency is above a certain level, any increase in efficiency is always a compromise between efficiency and cost.

Fig 9 to Fig 14 shows the fluid flow and fluid leakage flow path for various configurations. Letter "C" indicates a region where the fluid is in a compressed state and letter "A" where the fluid is at its lowest pressure state.
Fig 9 is a combination of PCT/GB01/03089 modified by incorporation of a slide ring and US 3895609 modified by making the side disc "L" shaped. Fig 9 shows the condition when the charge pressure is higher than the pressure in the machine. Fluid leaking into the machine can flow down the end clearance and directly into the working volume through the axial slots in the side disc. The other entry is over the relatively long diametral clearance of the extended "L" shaped disc and over the wide disc opposite via the pressure equalising transfer holes.
Fig 10 is as Fig 9 but when the machine is discharging. Leakage to the lower pressure region is over the relatively long diametral clearance on both discs.
Fig 11 and Fig 12 shows a machine similar to PCT/GB01/03089 except there is a slide ring for evacuation and control of mass flow. In both Fig 11 and Fig 12 leakage is over the relatively long diametral clearance on both discs.
Fig 13 and Fig 14 is similar to US 3895609 but modified by incorporating evacuation controlled by a slide ring. It will be appreciated that leakage will be significant over the relatively narrow side discs and lengthening the discs increases the transfer loss. However this design may be beneficial for controlling air flow in a low pressure blower.

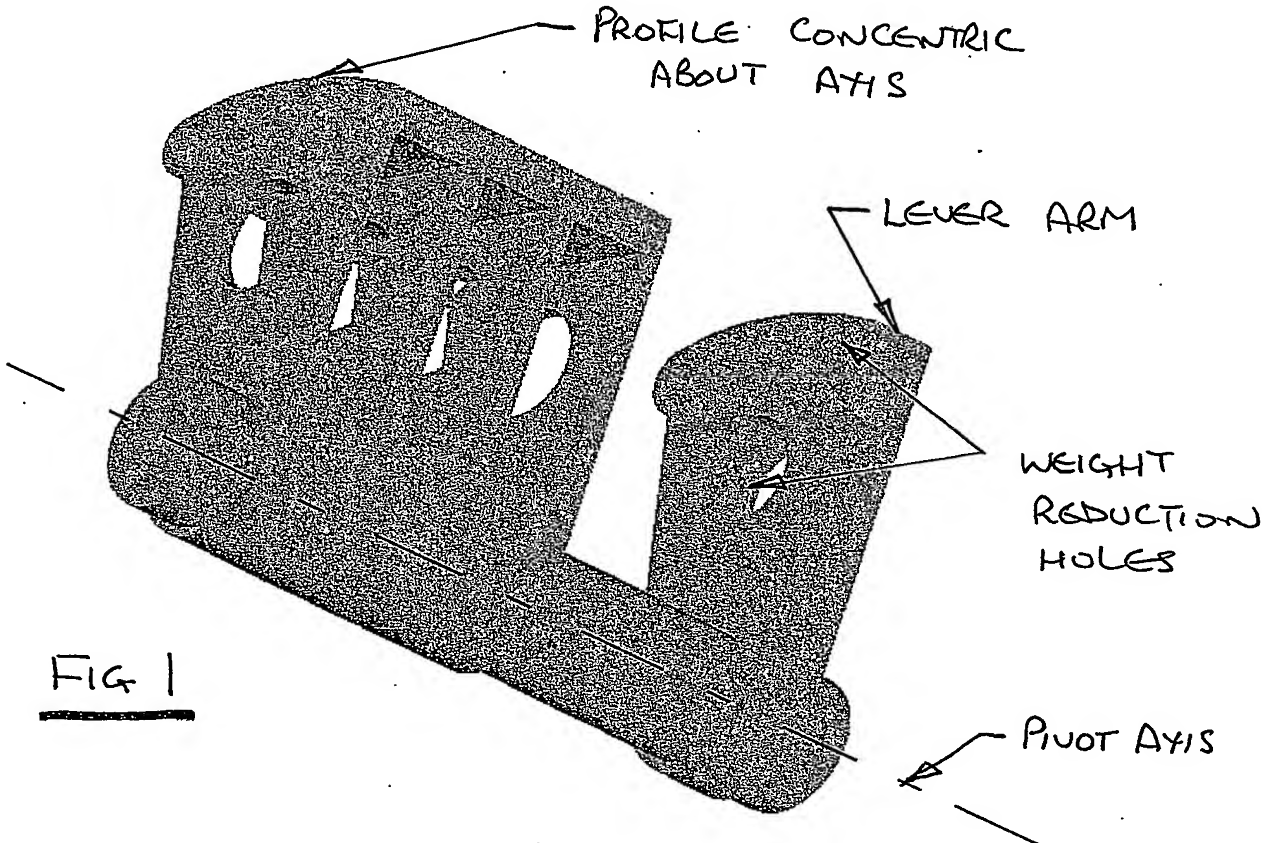
PROFILE CONCENTRIC ABOUT AXIS
LEVER ARM
WEIGHT REDUCTION HOLES
PIVOT AXIS

FIG 1

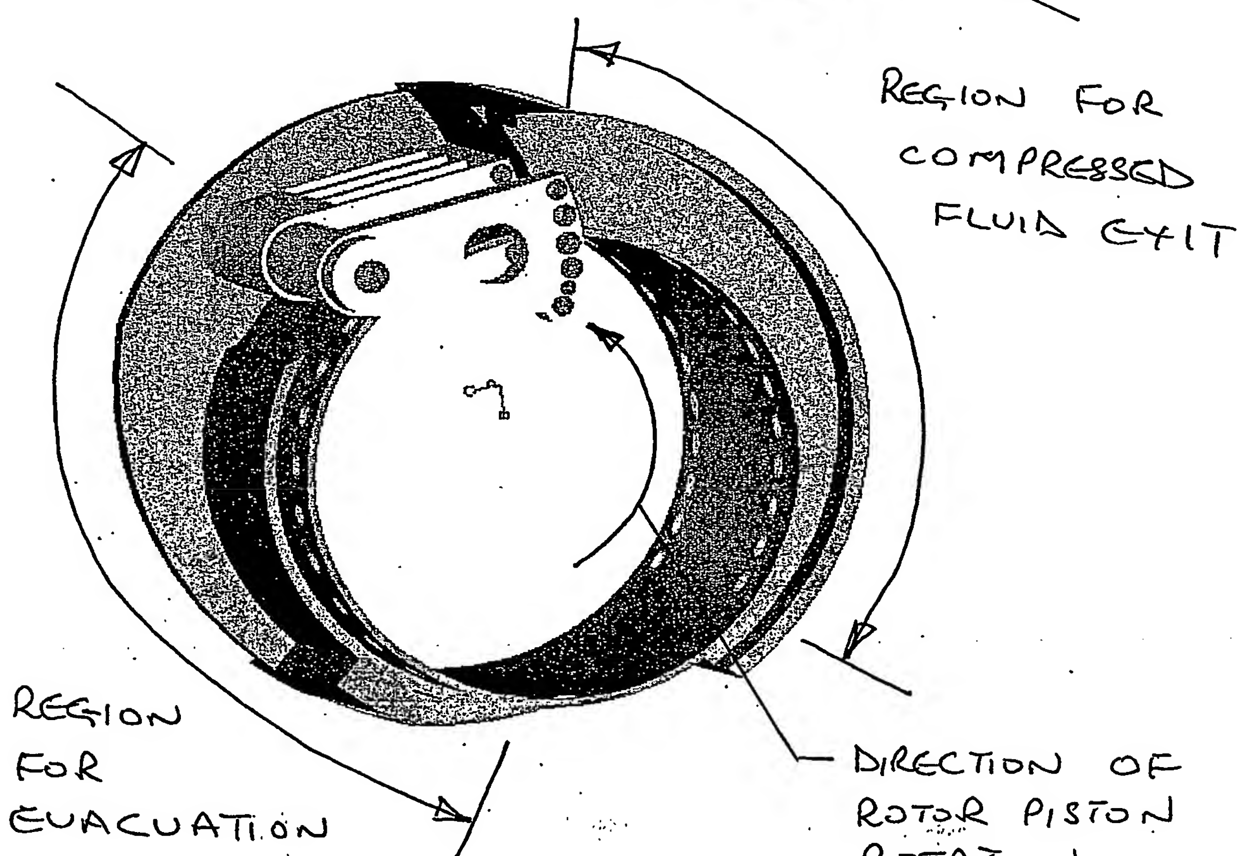
REGION FOR COMPRESSED FLUID EXIT
REGION FOR EVACUATION
DIRECTION OF ROTOR PISTON ROTATION

FIG 2

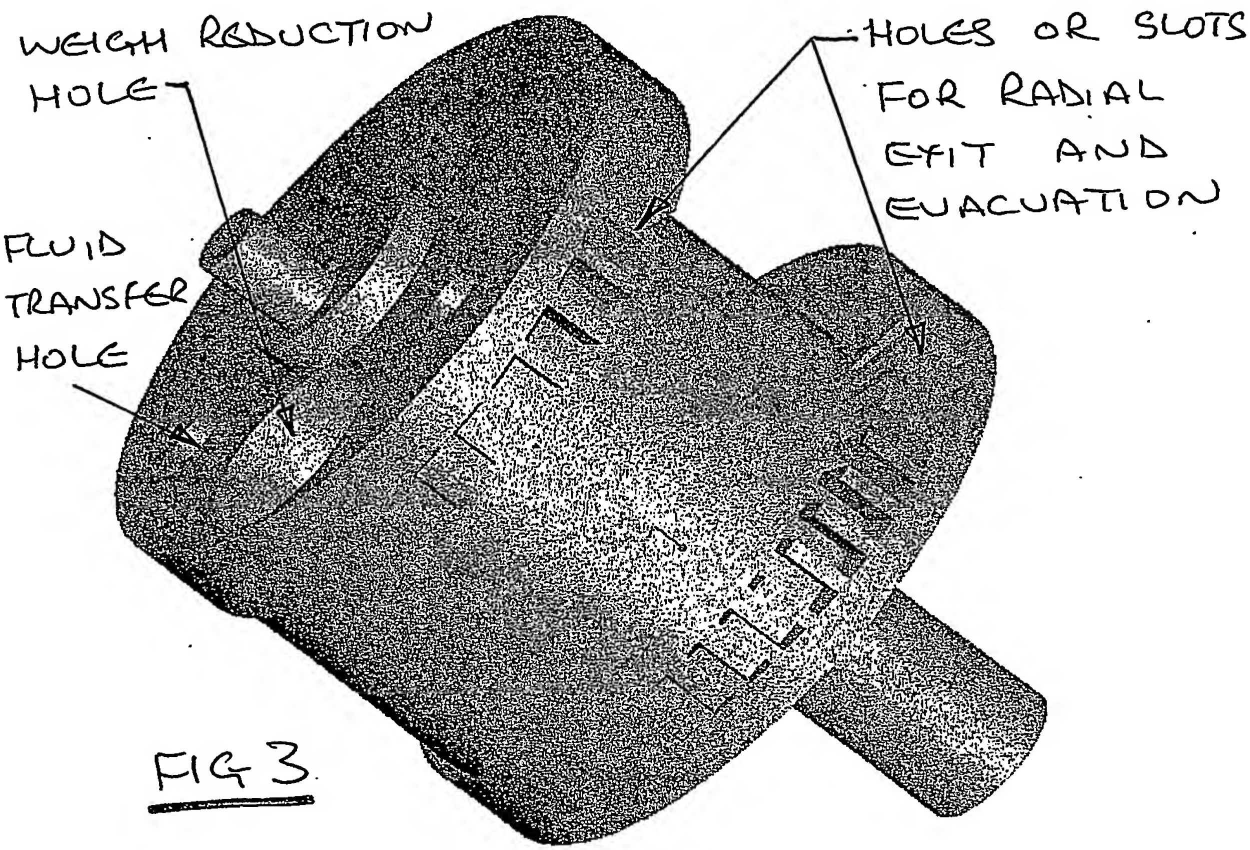
WEIGH REDUCTION HOLE
HOLES OR SLOTS FOR RADIAL EXIT AND EVACUATION
FLUID TRANSFER HOLE

FIG 3

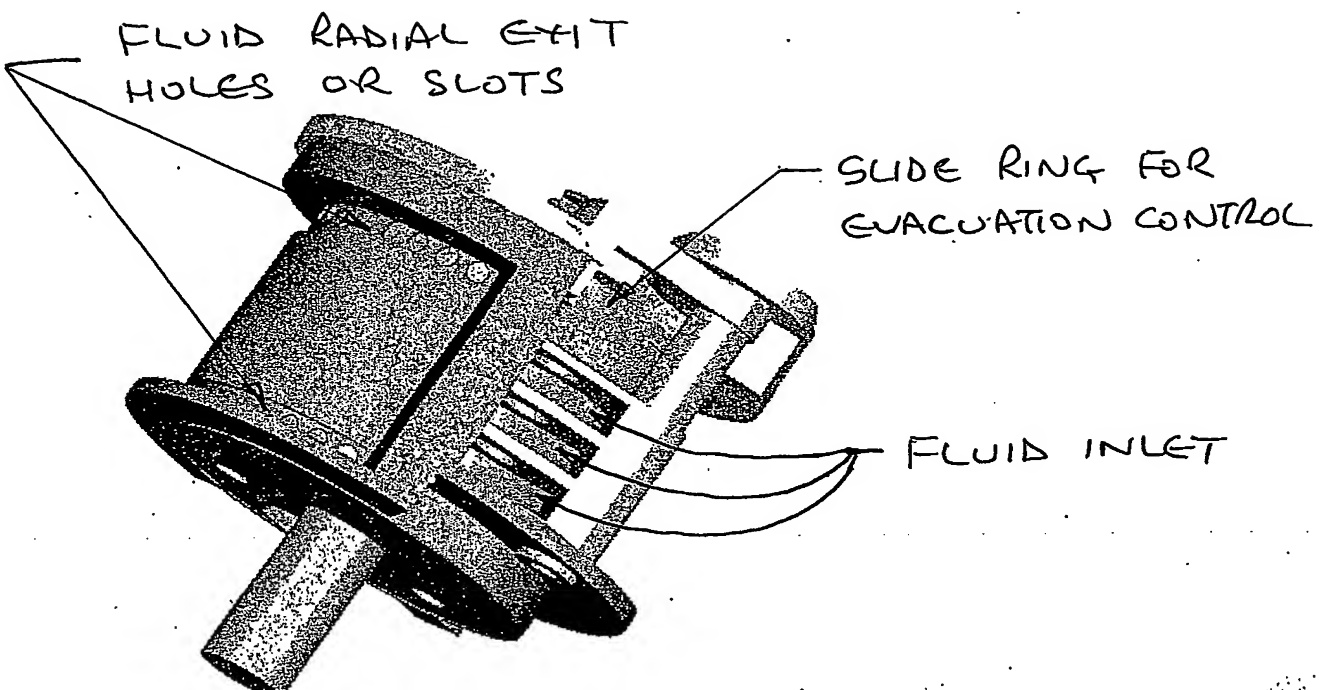
FLUID RADIAL EXIT HOLES OR SLOTS
SLIDE RING FOR EVACUATION CONTROL
FLUID INLET

FIG 4

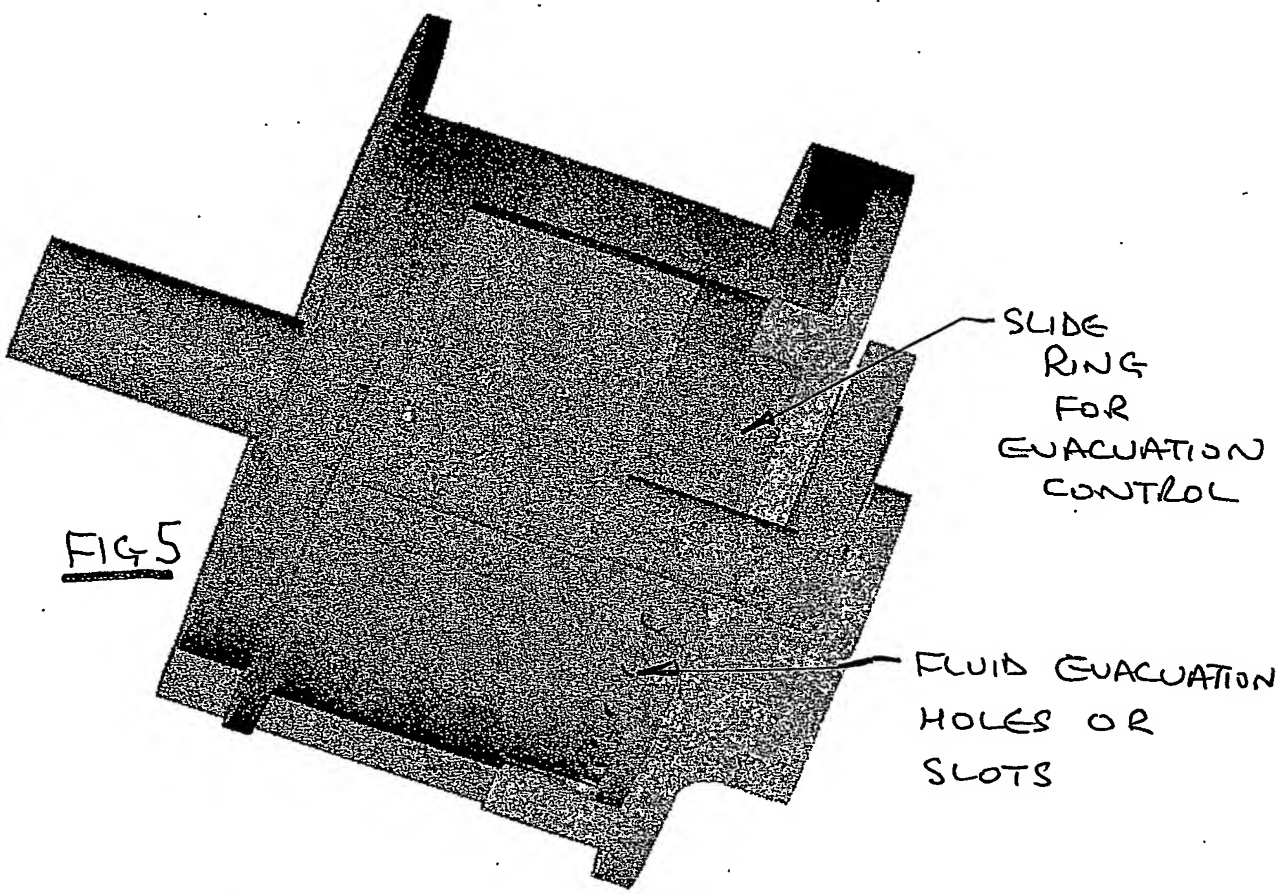
SLIDE RING FOR EVACUATION CONTROL
FLUID EVACUATION HOLES OR SLOTS

FIG 5

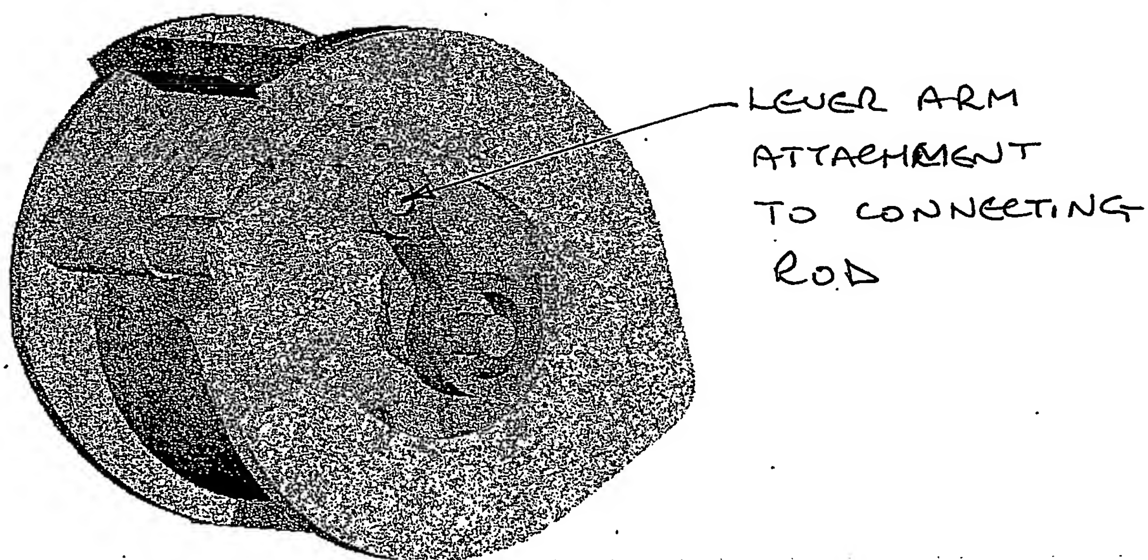
LEVER ARM ATTACHMENT TO CONNECTING ROD

FIG 6

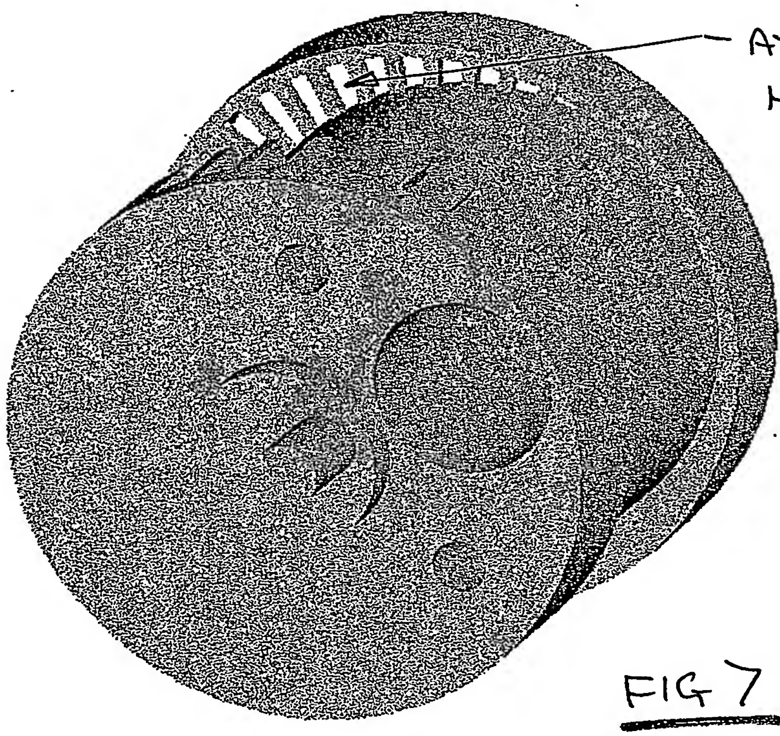

FIG 7

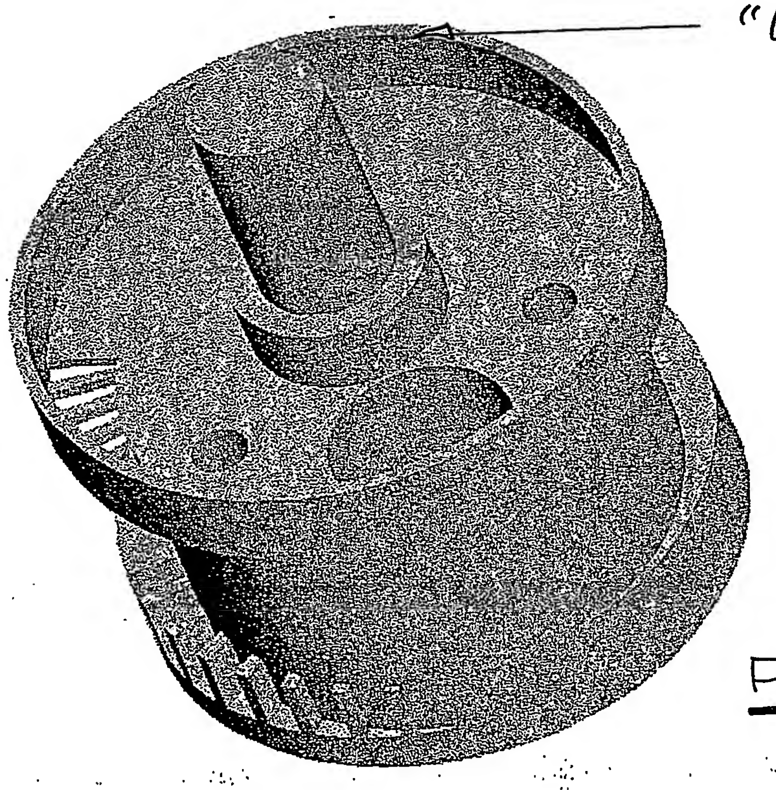

FIG 8

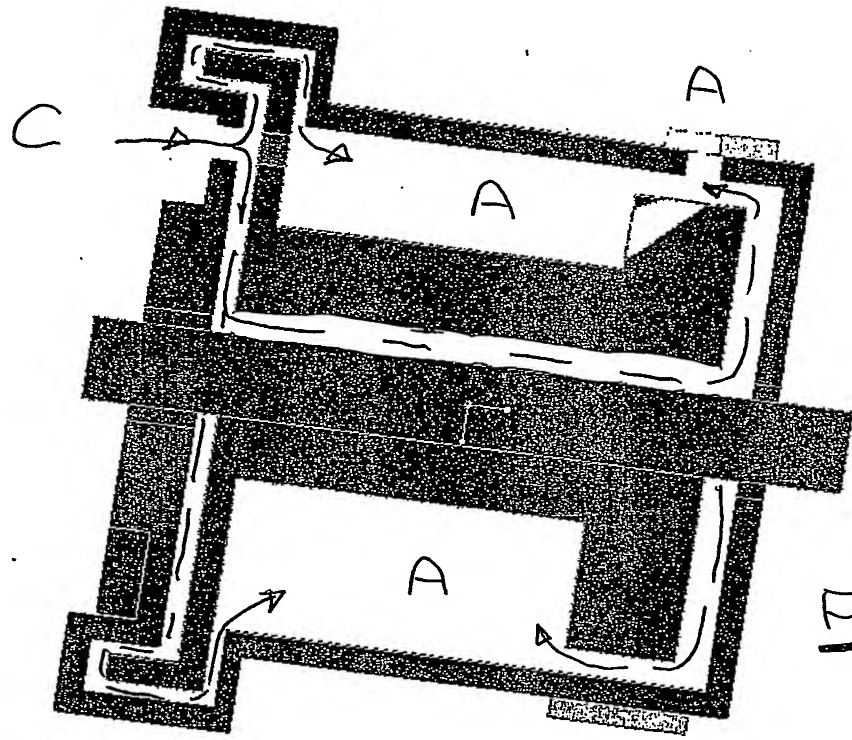

FIG 9

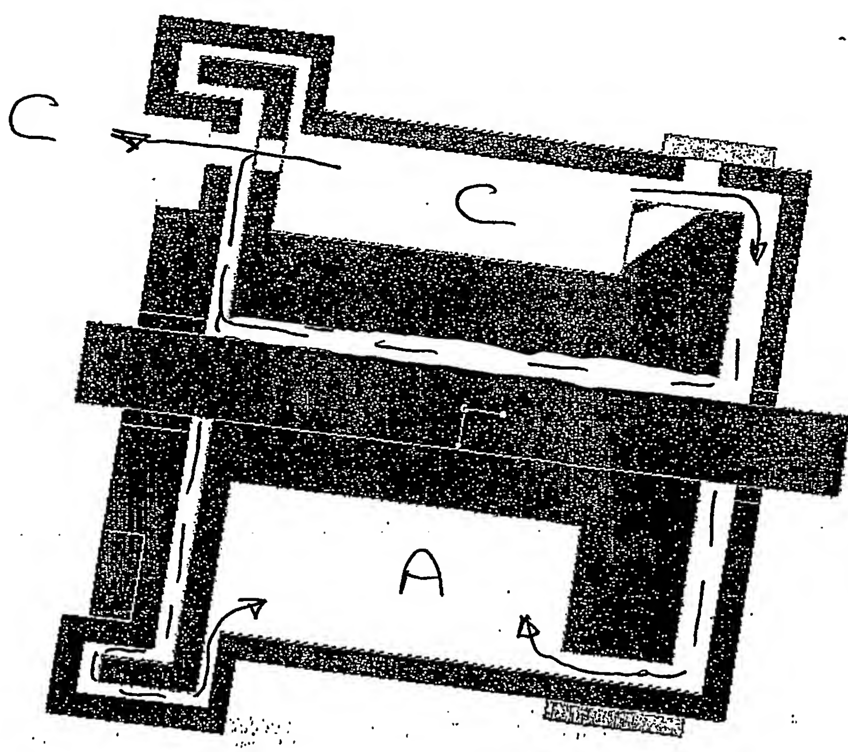

FIG 10

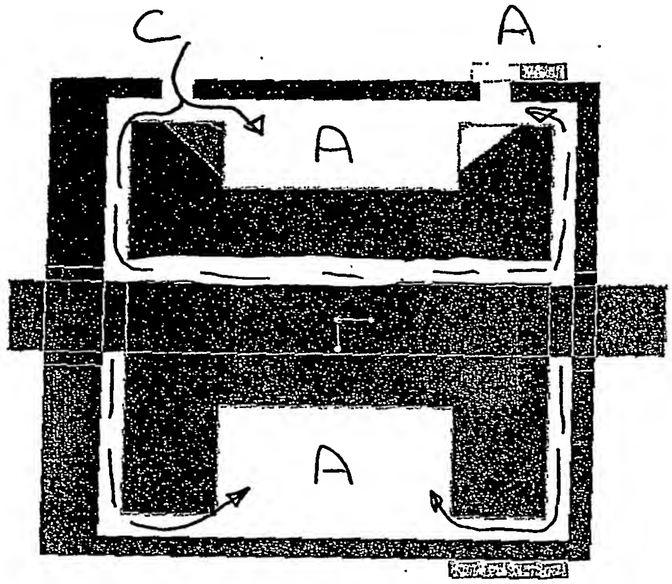

FIG 11

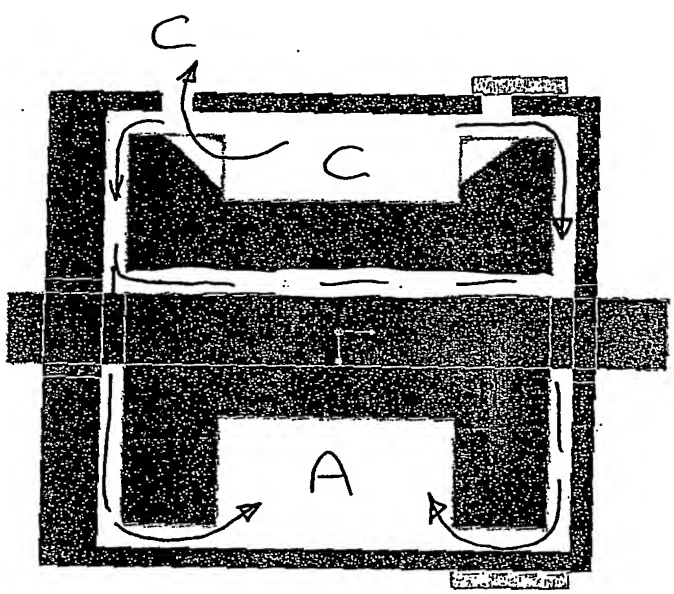

FIG 12

This Page is Inserted by IFW Indexing and Scanning Operations and is not part of the Official Record

BEST AVAILABLE IMAGES

Defective images within this document are accurate representations of the original documents submitted by the applicant.

Defects in the images include but are not limited to the items checked:

- ☐ BLACK BORDERS
- ☐ IMAGE CUT OFF AT TOP, BOTTOM OR SIDES
- ☐ FADED TEXT OR DRAWING
- ☐ BLURRED OR ILLEGIBLE TEXT OR DRAWING
- ☐ SKEWED/SLANTED IMAGES
- ☐ COLOR OR BLACK AND WHITE PHOTOGRAPHS
- ☐ GRAY SCALE DOCUMENTS
- ☐ LINES OR MARKS ON ORIGINAL DOCUMENT
- ☐ REFERENCE(S) OR EXHIBIT(S) SUBMITTED ARE POOR QUALITY
- ☐ OTHER: ______________________

IMAGES ARE BEST AVAILABLE COPY.
As rescanning these documents will not correct the image problems checked, please do not report these problems to the IFW Image Problem Mailbox.